Amendments to the Drawings

There are no amendments to the drawings.

Remarks/Arguments

In the specification, the line 26 of page 3 was amended deem that the density of 3.0 lbs. per cubic foot is critical to the design and significant. The correct density allows for proper condensing of the material to provide appropriate perturbations, thus allowing the invention to produce a desirable proprioceptive feedback for rehabilitation. The density of the foam is also critical in that it allows for a light weight, easily portable device.

The paragraph beginning on line 1 of page 8 was amended to correct informalities, and assure that there was consistency with reference characters in reference to "flat narrow surface 18".

In the Summary of the Invention (Disclosure), page 4, lines 7-17, were amended in order to make certain the fact that the dimensions of major diameter of 13.5 inches, minor diameter of 6.0 inches, and thickness of 3.0 inches is critical to the design and significant. These dimensions allow proper placement of one foot on the device while wearing a shoe. The thickness allows the ankle to reach all normal ranges of motion due to the distance of the first surface from the second when working in a tri-planar motion. Achieving normal range of motion at the ankle is critical in the proper rehabilitation of a patient. Those normal ranges being: Dorsiflexion: 0-20 degrees Plantarflexion : 0-50 degrees Inversion: 0-35 degrees Eversion: 0-15 degrees

Claims 1-2, claims 4-9, claims 11-15, claim 17 and claim 20 have been canceled.

Claim 3 has been amended to demonstrate that the major diameter, minor diameter and thickness are critical to the design and significant.

Claim 10 has been amended to demonstrate that the density of the foam is critical to the design and significant.
The examiner has acknowledged that claims 16-18 would be allowable if rewritten in independent form. Therefore, these claims were rewritten as instructed.

Claim19 has been amended to as following recommendations from claims 16-18.

Applicant respectfully requests that a timely Notice of Allowance be issued in this case.

Respectfully submitted,

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(54) Bearing/seal member/assembly and mounting

(57) A gas dynamic machine, such as a turbocharger, compressor or turbine, includes a housing and a rotor mounted in the housing and rotatable on an axis. A bearing and seal member or assembly includes a bearing mounted in an axial bore of the housing and having an inner bearing surface radially supporting a bearing journal of the rotor An axial oil seal is aligned with and fixed to the bearing for radially positioning the seal in axial alignment with the journal bearing surface of the bearing/seal assembly. The seal may further include a generally radial flange fixed to the housing for axially positioning the bearing/seal assembly relative to the rotor. The bearing and seal may have separate bodies attached together or be combined in an Integral body. Mounting with a single radial pilot for radial location and a single radial flange for axial location, which may be combined, is featured. An optional thrust flange may be integral with the bearing portion or attached as a separate body.

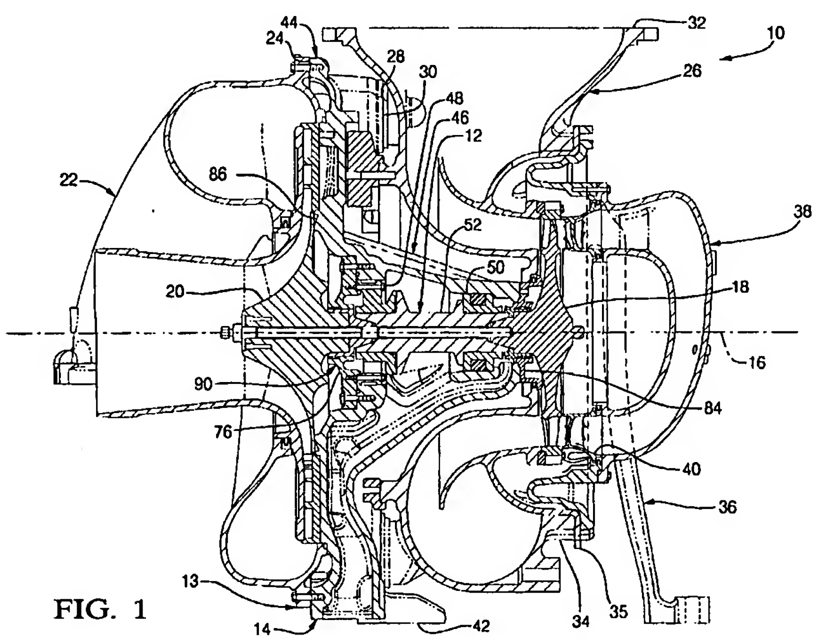

FIG. 1

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Description

TECHNICAL FIELD

[0001] This invention relates to gas dynamic machines, such as turbochargers, compressors, turbines and the like and, more particularly, to a bearing and seal member or assembly for and in combination with a gas dynamic machine.

BACKGROUND OF THE INVENTION

[0002] It is known in the art relating to gas dynamic machines, such as turbochargers, compressors, turbines, and the like, to provide a rotor supported in a housing on one or more oil lubricated bearings. An axial seal may be provided adjacent the bearing to control leakage of oil from the bearing into a compressor or turbine of the machine. Axial alignment of the seal with the bearing is generally required to provide proper seal operation. To accomplish this, concentric pilots or bores may be provided in the machine housing, a bearing being mounted in one of the bores and a seal member mounted in the other. The result is that close machine tolerances must be held in both the bearing and seal components and the pilot bores of the housing in order to obtain the desired concentricity. Installation and removal of the components may also be complicated and may require special tooling for servicing of the machine. An improved bearing and seal mounting was desired to improve the operation, manufacture installation and servicing of rotor bearings and seals for a gas dynamic machine.

SUMMARY OF THE INVENTION

[0003] The present invention provides a new bearing/seal member or assembly and a modified mounting in a gas dynamic machine, which accomplish the above-mentioned goals in an engine turbocharger and other similar machines.

[0004] In a first embodiment, the bearing and seal portions of the assembly are made as separate bodies provided with mating alignment portions that maintain axial alignment of the bearing and seal surfaces when the components are fixed together as an assembly. The assembly is radially positioned by a single pilot bore of the housing in which the bearing portion is mounted. Concentricity of the bearing and seal portions is thus determined only by the alignment portions of the bearing and seal bodies themselves and does not depend upon the tolerances of dual housing pilot bores. The axial positioning of the assembly is also determined by a single flange formed, in this case, as part of the seal body. Preferably, a thrust flange or thrust bearing surface is formed integrally on the bearing body, axially aligned with the bearing surface on an end opposite the seal body. If desired, the thrust flange could be separately mounted on the bearing body.

[0005] In use, the bearing and seal bodies are assembled with their bearing and seal surfaces in alignment and fixed together by fasteners prior to installation in the turbocharger housing. The assembly is then assembled into the housing with a pilot surface of the bearing portion supported in the housing pilot bore. The radial flange of the seal portion is then secured to a radial mounting face of the housing by, for example, screw fasteners to retain the assembly in place. The bearing/seal assembly is easily installed and may be easily removed without special tooling.

[0006] In an alternative embodiment, the bearing and seal portions are formed integrally in a single body. The body includes a single radial mounting flange, which carries a peripheral external pilot surface on the flange for engaging a pilot bore of a turbocharger housing to radially position the body in the housing. The mounting flange also is engagable with the housing for axially positioning and mounting the body in the housing. Preferably, a thrust flange having a thrust bearing surface is separately formed and attached to the bearing/seal body at the end adjacent the bearing portion. If desired, the thrust bearing surface could be formed integral with the bearing/seal body. The bearing portion of the body includes an inner bearing surface in which a floating bearing bushing with squeeze film damping is preferably received. Alternatively, a fixed bearing insert or a directly formed bearing material could be carried by the inner bearing surface.

[0007] These and other features and advantages of the invention will be more fully understood from the following description of certain specific embodiments of the invention taken together with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0008] In the drawings:

FIG. 1 is a cross-sectional view of an engine turbocharger having a single pilot mounted bearing/seal assembly according to the invention;

FIG. 2 is an enlarged view of the bearing/seal mounting portion of FIG. 1;

FIG. 3 is an exploded pictorial view of the bearing/seal assembly of FIGS. 1 and 2;

FIG. 4 is a pictorial view of the assembled bearing/seal assembly.

FIG. 5 is a fragmentary cross-sectional view of a modified turbocharger having an integrated bearing/seal member with attached thrust bearing according to the invention; and

FIG. 6 is an exploded pictorial view of the bearing/seal member of FIG. 5.

DESCRIPTION OF THE PREFERRED EMBODIMENT

[0009] Referring now to the drawings in detail, numeral 10 generally indicates an exhaust driven turbocharger for an engine, such as a diesel engine intended for use in railway locomotives or other applications of medium speed diesel engines. Turbocharger 10 includes a rotor 12 carried in a housing 13 by a rotor support 14 for rotation on a longitudinal axis 16 and including a turbine wheel 18 and a compressor wheel 20. The compressor wheel is enclosed by a compressor housing assembly 22 including components which are supported on an axially facing first side 24 of the rotor support 14. An exhaust duct 26 has a compressor end 28 that is mounted on a second side 30 of the rotor support 14 spaced axially from the first side 24.

[0010] The exhaust duct 26 is physically positioned between the rotor support 14 and the turbine wheel 18 to receive exhaust gases passing through the turbine wheel and carry them to an exhaust outlet 32. A turbine end 34 of the exhaust duct 26 and an associated nozzle retainer assembly 35 are separately supported by an exhaust duct support 36 that is connected with the exhaust duct 26 at the turbine end 34. The exhaust duct support 36 also supports a turbine inlet scroll 38 which receives exhaust gas from the associated engine and directs it through a nozzle ring 40 to the turbine wheel 18 for transferring energy to drive the turbocharger compressor wheel 20. The rotor support 14, compressor housing assembly 22, exhaust duct 26, nozzle retainer assembly 35 and exhaust duct support 36 are all included as portions of the housing 13 of the turbocharger 10 that represents one example of a gas dynamic machine according to the invention.

[0011] The rotor support 14 includes a pair of laterally spaced mounting feet 42 which are rigidly connected to an upstanding mounting portion 44 of the rotor support 14 and are adapted to be mounted on a rigid base, not shown. The rotor support 14 further includes a tapering rotor support portion 46 having bearings 48, 50 that rotatably support the rotor 12. Bearing 48 is a combination journal and thrust bearing, while bearing 50 is primarily a journal bearing.

[0012] The rotor 12 includes a shaft 52 connected with the turbine wheel 18 at one end and the compressor wheel 20 at the opposite end. The shaft 52 includes a pair of axially spaced bearing journals 54, 56 respectively adjacent the compressor and turbine wheel ends of the shaft. A flange, inboard of journal 54, carries a radial thrust reaction surface 58. A second flange adjacent journal 56 carries a radial surface 60. Journals 54, 56 are respectively supported in bearings 48, 50. Radial surface 58 carries thrust forces to the journal/thrust bearing 58 and radial surface 60 limits axial movement of the rotor.

[0013] Connecting means of any suitable type may be provided for aligning and connecting the compressor and turbine wheels on their respective ends of the shaft 52. The aluminum alloy compressor wheel 20 includes an axially aligned cylindrical stub 62 on which is fixed an adapter 64 including an outer seal surface 65. For the compressor wheel 18, the connecting means comprise a pair of non-locking cones between the adapter 64 and the shaft 52. For the turbine wheel 18, the connecting means include non-locking cones between the turbine wheel and the shaft 52. A seal collar 66 fixed on the turbine wheel adjacent the cones includes an outer seal surface 68. The rotor elements are secured together by fastener means including a nut 70 and a long stud 72, or a bolt, that extends through the compressor wheel 20 and shaft 52 to engage the turbine wheel 18. The stud and nut hold the non-locking cones in engagement to maintain the compressor and turbine wheels in axial alignment on the shaft 52. Suitable mechanical stops may be provided between the shaft and wheel elements of the rotor to provide angular orientation and allow reassembly of the elements in predetermined angular relation.

[0014] The outer seal surface 65 of adapter 64 is an outer cylinder located between the compressor wheel 20 and bearing journal 54, which is supported by oil lubricated bearing 48 (FIG. 1). The outer cylinder 65 is surrounded by an inner seal surface or cylinder 74, formed by a bore in a compressor seal 76 having a radial mounting flange 78 fixed to a radial mounting face 80 of the housing 13 at the compressor end of the support portion 46. Similarly, the outer seal surface 68 of seal collar 66 is an outer cylinder located between the turbine wheel 18 and bearing journal 56, which is supported by oil lubricated bearing 50. The outer cylinder 68 is surrounded by an inner seal surface or cylinder 92, formed by a bore in a turbine seal 84 fixed to the rotor support 14 at the compressor end of the support portion 46. The outer cylinders 65, 68 are centered within the inner cylinders 74, 82 with a predetermined close clearance selected to enhance sealing action of dual phase seals partially defined by opposing cylinders 65, 68 and 74, 82.

[0015] The outer cylinders 65, 68 are each provided with auger seal grooving consisting of a multi-start helical thread cut into the outer cylinders 65, 68. The threads lie opposite smooth bore portions on the inner ends of the associated inner cylinders 74, 82. The threads have helix angles turning in opposite directions, chosen so that rotation of the rotor causes a viscous pumping action of the threads against the smooth bores that forces oil entering the clearance back toward the associated bearings.

[0016] The inner cylinders 74, 82 are each provided with labyrinth seal grooving consisting of spaced circumferential lands and grooves cut into the outer ends of the inner cylinders 74, 82. The labyrinth seal grooving lies opposite smooth surfaced portions of the associated outer cylinders 65, 68. A central groove receives air pressure through passages in the compressor seal 76 and the turbine seal 84. The air pressure is received

from an annular groove 86 in the turbocharger rotor support mounting portion 44 (FIG. 1) behind the back face of the compressor wheel near its outer periphery. The air pressure is carried through internal passages and distributed across the clearance from the smooth surfaced portions of the outer cylinders 65, 68 and partially flows back through the clearance toward the adjacent bearings 48, 50, further preventing the passage of oil through the clearance toward the compressor and turbine wheels. The complementary auger seals and labyrinth seals in the cooperating cylinders provide non-rubbing seal assemblies as used in the described turbocharger. However other suitable forms of axial seals, whether non-rubbing or not, could also be applied within the scope of the present invention.

[0017] In accordance with the invention, the bearing 48 and the compressor seal 76 are fixed together in a bearing/seal assembly 90 by suitable fasteners in the form of six screws 92. The bearing 48 includes a body 94 having an inner bearing surface 96 surrounding and radially supporting the bearing journal 54 of the rotor shaft 52. A radial mounting flange 98 is provided having an outer mounting surface including first and second circular or generally cylindrical pilot portions 100, 102 separated by an oil distribution groove 104 for supplying oil to the bearing surface 96. A thrust flange extends from one end of the bearing body 94 and includes a thrust bearing surface 106 engagable with reaction surface 58 of the rotor shaft 52. At its other end, the body 94 includes a seal aligning portion 108 including radial end and internal cylindrical surfaces, not shown.

[0018] The compressor seal 76 includes a seal body 110 that carries the seal inner cylinder 74, previously described, which is connected with the radial flange 78. An inner portion of the flange 78 includes a radial face 112 with an inwardly adjacent cylindrical guide portion 114. These together form a mating aligning portion of the seal, which engages the seal aligning portion 108 of the bearing body 94 to insure close axial alignment of the seal inner cylinder 74 with the inner bearing surface 96 of the bearing body 94. An outer portion of the flange 78 includes a radial surface 116 that engages the mounting face 80 of the rotor support portion 46 of the turbocharger housing 13.

[0019] In use, the bearing 48 and seal 76 are separately manufactured to close tolerances. These members are then assembled with the seal aligning portion 108 of the bearing 48 engaging the mating aligning portions 112, 114 of the seal 76. Screws 92 are inserted from the inner side of the bearing 48 through holes in the flange 98 to engage threaded openings in the radial face 112 of the seal 76 and fix the components together in the bearing/seal assembly 90 of the invention. The members 48, 76 of the bearing/seal assembly 90 are thus fixed with inner bearing surface 96 and the inner cylinder 74 of the seal in close axial alignment.

[0020] Assembly 90 is then installed into the housing 13 from the compressor end. The generally cylindrical pilot portions 100, 102 of flange 98 are received in a close fitting generally cylindrical bore 118 in the rotor support portion 46 of the housing 13. The radial flange 78 of the seal body is not closely fitted radially within the housing 13. Instead, the seal body 110 relies upon the mounting of the bearing body 94 of bearing 48 for radial support and alignment. The diameter of the bore 118 may vary slightly at its opposite ends to assist installation and to mate closely with the pilot portions 100, 102, of which the inner pilot portion 100 is slightly smaller in diameter. The close fit of the mounting flange 98 in the bore 118 insures the axial alignment of the bearing/seal assembly 90 with the axis of the rotor 12 and housing 13.

[0021] The bearing/seal assembly 90 is axially fixed in the housing 13 by screws 120 installed from the outer side through holes in the seal flange 78 to engage threaded openings in the rotor support portion 46 of the housing. The flange 78 is thus fixed against the mounting face 80 to axially position both the bearing 48 and the seal 76 in the housing 13.

[0022] Referring to FIG. 5, the compressor bearing mounting portion of a modified engine turbocharger 130 is shown having features similar to those of turbocharger 10 and in which like numerals are used for like parts. Turbocharger 130 differs primarily in the structure of a compressor bearing/seal member 132, which forms a part of the present invention, and the drive arrangement for the rotor 134, which does not. The rotor includes the compressor wheel 136 and turbine wheel, not shown, connected by a shaft 140, all similar to components of rotor 12. However, instead of a cone drive connection, rotor 134 includes mating toothed couplings 142, 144 on the shaft 140 and on an adapter 146 carried on a stub of the compressor wheel 136. Similar couplings are used between the shaft 140 and the turbine wheel, not shown. The adapter 146 also provides an outer seal surface 148 forming an outer cylinder having a smooth surface at the compressor end and auger seal grooving at the shaft end similar to the corresponding cylinder 65 of FIG. 2.

[0023] In accordance with the invention, the compressor bearing/seal member 132 includes an integral body 150 having a bearing portion 152 at one end and a seal portion 154 at the other end. The bearing portion 152 includes an inner bearing surface 156 fed by internal oil passages. A floating bearing bushing 158 incorporating a squeeze film damper is received in the inner bearing surface for supporting the associated shaft bearing journal 54. However, other bearing arrangements, such as sleeve bearings or integrated bearing materials, could be used with the inner bearing surface in suitable turbocharger applications.

[0024] The seal portion 154 includes a seal inner cylinder 160 (FIG. 5) including labyrinth seal grooving at the compressor wheel end and a smooth cylindrical surface at the shaft end as in the previously described embodiment. It should be understood that other forms of seal surfaces could be alternatively provided in various

embodiments within the scope of the invention.

[0025] A single radial mounting flange 162 extends outward from the integral body 150 and includes a radial mounting face 164 and a cylindrical outer pilot surface 166. Pilot surface 166 is engagable with an inner pilot locating surface of the turbocharger housing for radially positioning and supporting the bearing/seal member in the housing. The radial mounting face 164 of flange 162 is engagable with the radial mounting face 80 of the turbocharger housing 13 to locate the bearing/seal member 132 and the supported rotor 134 axially in the housing. The flange 162 is secured to the housing by screws 120 to support and locate the member 132 as in the previously described embodiment.

[0026] The bearing/seal member 132 of the invention also optionally includes a thrust bearing in the form of an annular ring or flange 168 having a thrust bearing surface 170. The flange 168 is aligned by pins 172 with the shaft end of the bearing portion 152 of member 132. When assembled, the thrust flange 168 is axially aligned with the inner bearing surface 156 of the bearing portion and the inner cylinder 160 of the seal portion. If desired, the thrust flange 168 could be made as an integral part of the bearing/seal body 150. However, the preferred separate flange 168 reduces manufacturing complexity. Conversely, integration of the bearing and seal portions of the bearing/seal member 132 into a single body 150 simplifies the manufacture and assembly of these portions.

[0027] As installed in a turbocharger, the bearing/seal member 132 with associated thrust flange 168 functions in the same manner as the bearing/seal assembly 90 first described, except for the function of the squeeze film damper bearing bushing 158. This bushing supports the shaft bearing journal 54 while the thrust flange accepts thrust loads from the thrust reaction surface 58. The seal inner cylinder 160 also cooperates with outer seal surface 148 of adapter 146 to control oil leakage from the bearing portion 152 of the body 150.

[0028] It should be understood that, in its broader aspects, the invention is not limited by the particular form of axial positioning of the bearing or the specific manner of alignment or fastening of the components of the bearing/seal assembly. Also, the placement of the single pilot surface on the radial mounting flange or on a separate surface of the bearing/seal member or assembly may be varied to suit the application. The types of support bearings and lubricant seals used may also be varied, although the invention is particularly suitable for the embodiments described.

[0029] The use of a pre-assembled bearing/seal assembly or an integral bearing/seal member as described, provides accurate alignment of the bearing and seal elements in a rotor housing with a reduction in manufacturing tolerances between the assembly and the surrounding machine housing. Thus, quality may be improved and manufacturing costs reduced by the application of the invention in an appropriate apparatus.

[0030] While the invention has been described by reference to certain preferred embodiments, it should be understood that numerous changes could be made within the spirit and scope of the inventive concepts described. Accordingly, it is intended that the invention not be limited to the disclosed embodiments, but that it have the full scope permitted by the language of the following claims.

Claims

1. A bearing/seal member for supporting and sealing a rotor in a housing of a gas dynamic machine, said member comprising:

   a bearing portion having an inner bearing surface for surrounding and radially supporting a bearing journal of the rotor;
   a seal portion having an inner seal surface for surrounding an outer seal surface of the rotor; and
   a body carrying said bearing and seal portions with said inner bearing and seal surfaces in spaced coaxial relation, said body including an outer pilot surface for radially engaging an inner locating surface of the housing, and a generally radial mounting flange axially engagable with a mounting face of the housing for axially mounting and positioning the body in the housing.

2. A bearing/seal member as in claim 1 wherein said bearing and seal portions of the body are formed as an integral element.

3. A bearing/seal member as in claim 1 wherein said bearing and seal portions of the body are formed as separate elements and fixed together.

4. A bearing/seal member as in claim 1 wherein said outer pilot surface is formed on said radial flange.

5. A bearing/seal member as in claim 1 wherein said outer pilot surface is formed separate from said radial flange.

6. A bearing/seal member as in claim 1 including a thrust bearing surface at an end opposite from said inner seal surface.

7. A bearing/seal member as in claim 6 wherein said thrust bearing surface is formed on a thrust flange separate from but attached to said bearing portion.

8. A bearing/seal member as in claim 6 wherein said thrust bearing surface is formed integral with said bearing portion.

9. A bearing/seal member as in claim 1 wherein said inner bearing surface carries a bearing material that in assembly is fixed relative to the inner bearing surface.

10. A bearing/seal member as in claim 1 wherein said inner bearing surface carries a floating bushing bearing.

11. A bearing/seal member as in claim 1 wherein said outer pilot surface comprises the sole means for radially positioning said member in a housing.

12. A bearing/seal member as in claim 1 wherein said radial mounting flange comprises the sole means for axially positioning said member in a housing.

13. A bearing/seal member for supporting and sealing a rotor in a housing of a gas dynamic machine, said member comprising:

a bearing portion having an inner bearing surface for surrounding and radially supporting a bearing journal of the rotor;
a seal portion having an inner seal surface for surrounding an outer seal surface of the rotor; and
a body integrally carrying said bearing and seal portions with said inner bearing and seal surfaces in spaced coaxial relation, said body including a generally radial mounting flange axially engagable with a mounting face of the housing for axially mounting and positioning the body in the housing, and an outer pilot surface on said flange for radially engaging an inner locating surface of the housing.

14. A bearing/seal member as in claim 13 including a thrust bearing disposed at an end adjacent said bearing portion, said thrust bearing formed separately from said body and fixed thereto.

15. A bearing/seal member as in claim 13 wherein said outer pilot surface comprises the sole means for radially positioning said member in a housing.

16. A bearing/seal member as in claim 13 wherein said radial mounting flange comprises the sole means for axially positioning said member in a housing.

17. A bearing/seal assembly for supporting and sealing a rotor in a housing of a gas dynamic machine, said assembly comprising:

a bearing having a bearing body with an inner bearing surface for surrounding and radially supporting a bearing journal of the rotor, an outer pilot surface for engaging a supporting inner surface of the housing, and a seal aligning portion at one axial end of the bearing body; and
a seal having a seal body with an inner seal surface for surrounding an outer seal surface of the rotor, a mating aligning portion mated with the seal aligning portion of the bearing body, and a generally radial mounting flange axially engagable with a mounting face of the housing for axially mounting and positioning the assembly in the housing.

18. A bearing/seal assembly as in claim 17 wherein said outer pilot surface of the bearing body is generally circular and centered on an axis.

19. A bearing/seal assembly as in claim 18 wherein said seal aligning portion of the bearing body includes a generally circular aligning surface concentric with the outer pilot surface.

20. A bearing/seal assembly as in claim 19 wherein said mating aligning portion of the seal body includes a generally circular mating surface coaxial with and radially engaging the aligning surface of the bearing body for axially aligning the bearing and seal bodies and their inner bearing and seal surfaces.

21. A bearing/seal assembly as in claim 20 wherein said aligning portions of the bearing and seal bodies include generally radial axially engaging mating positioning surfaces for axially locating the bearing relative to the seal.

22. A bearing/seal assembly as in claim 17 wherein said bearing body further includes a thrust bearing surface at an end opposite from the seal aligning portion.

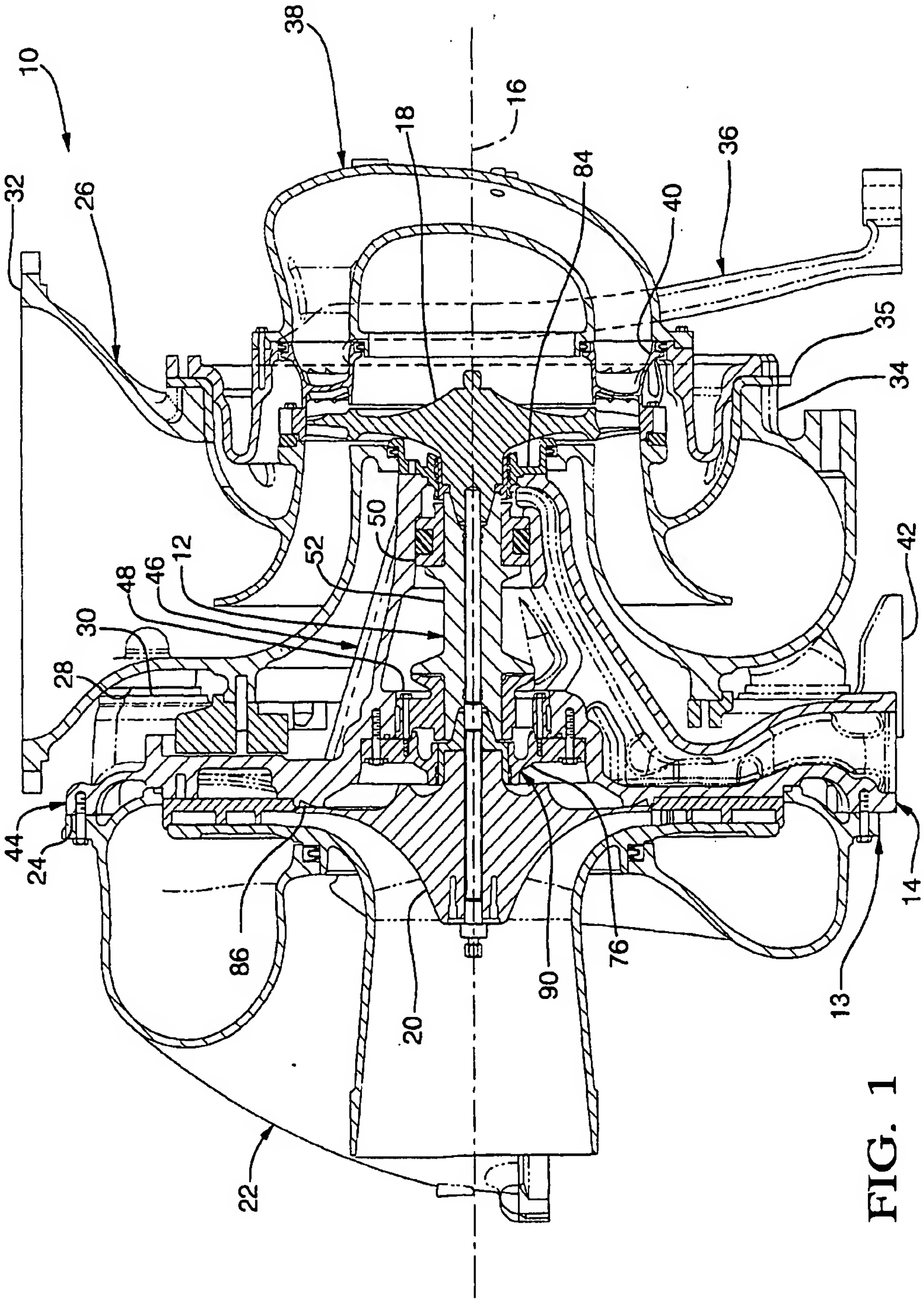
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FIG. 1

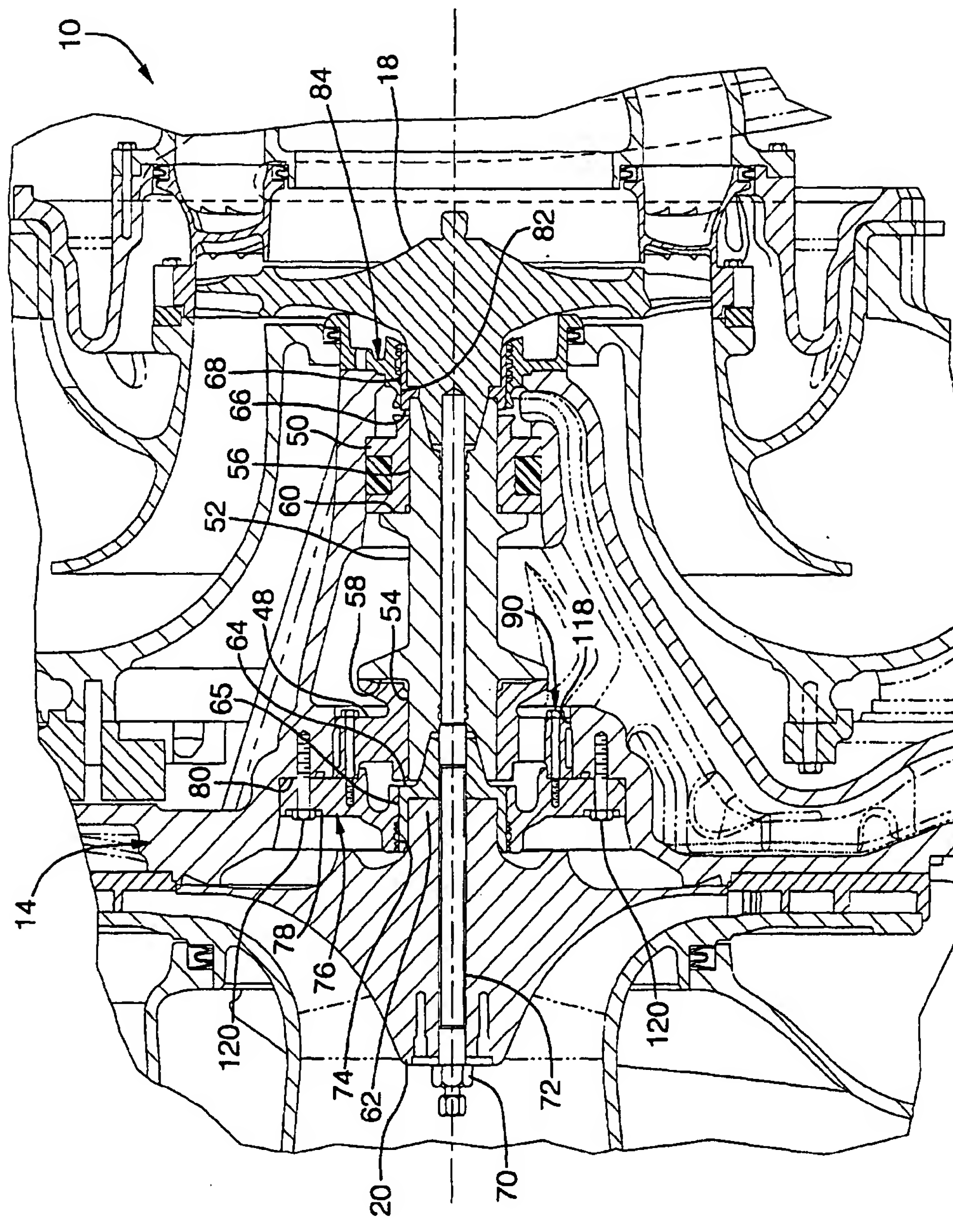
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FIG. 2

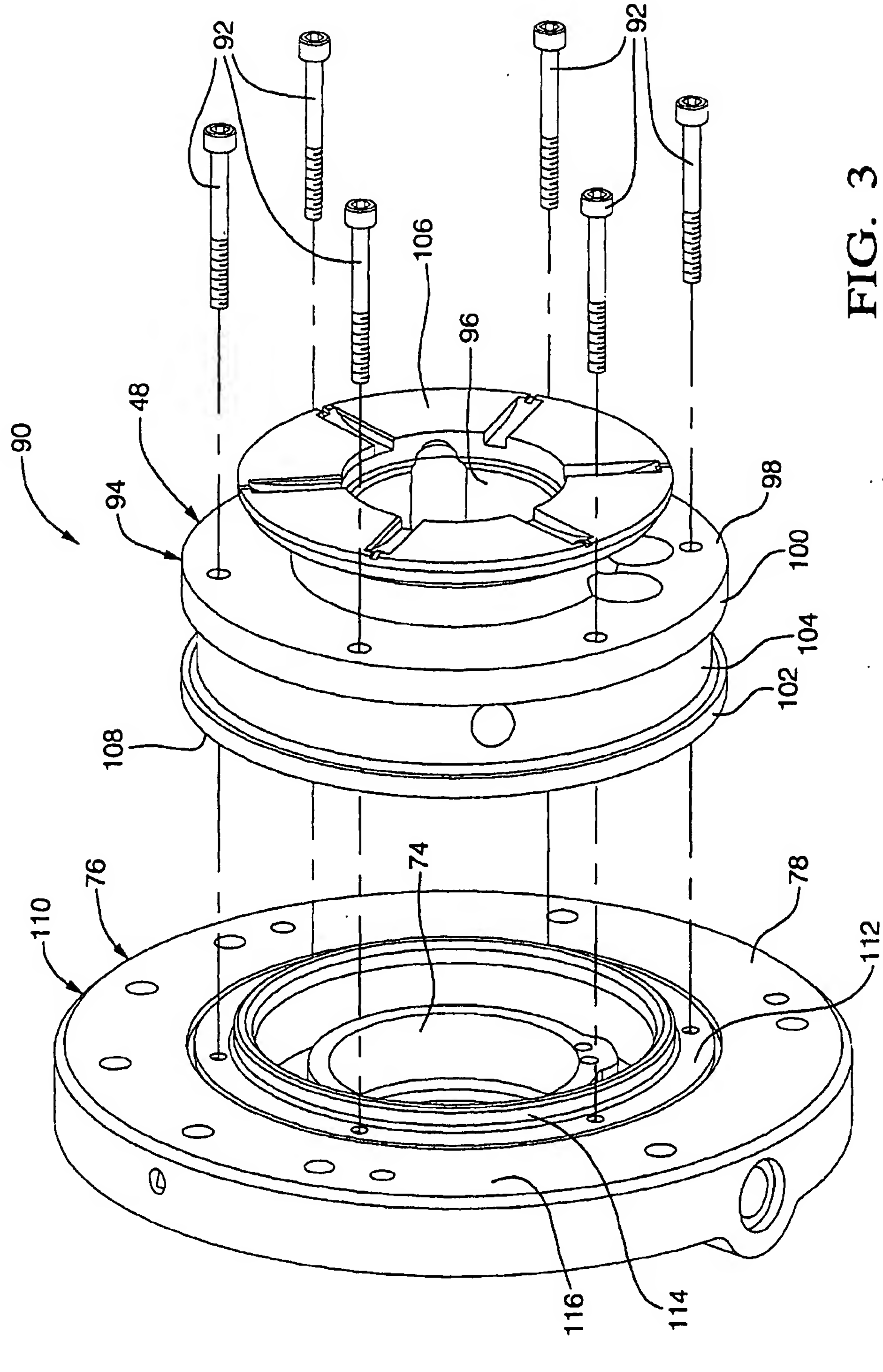

FIG. 3

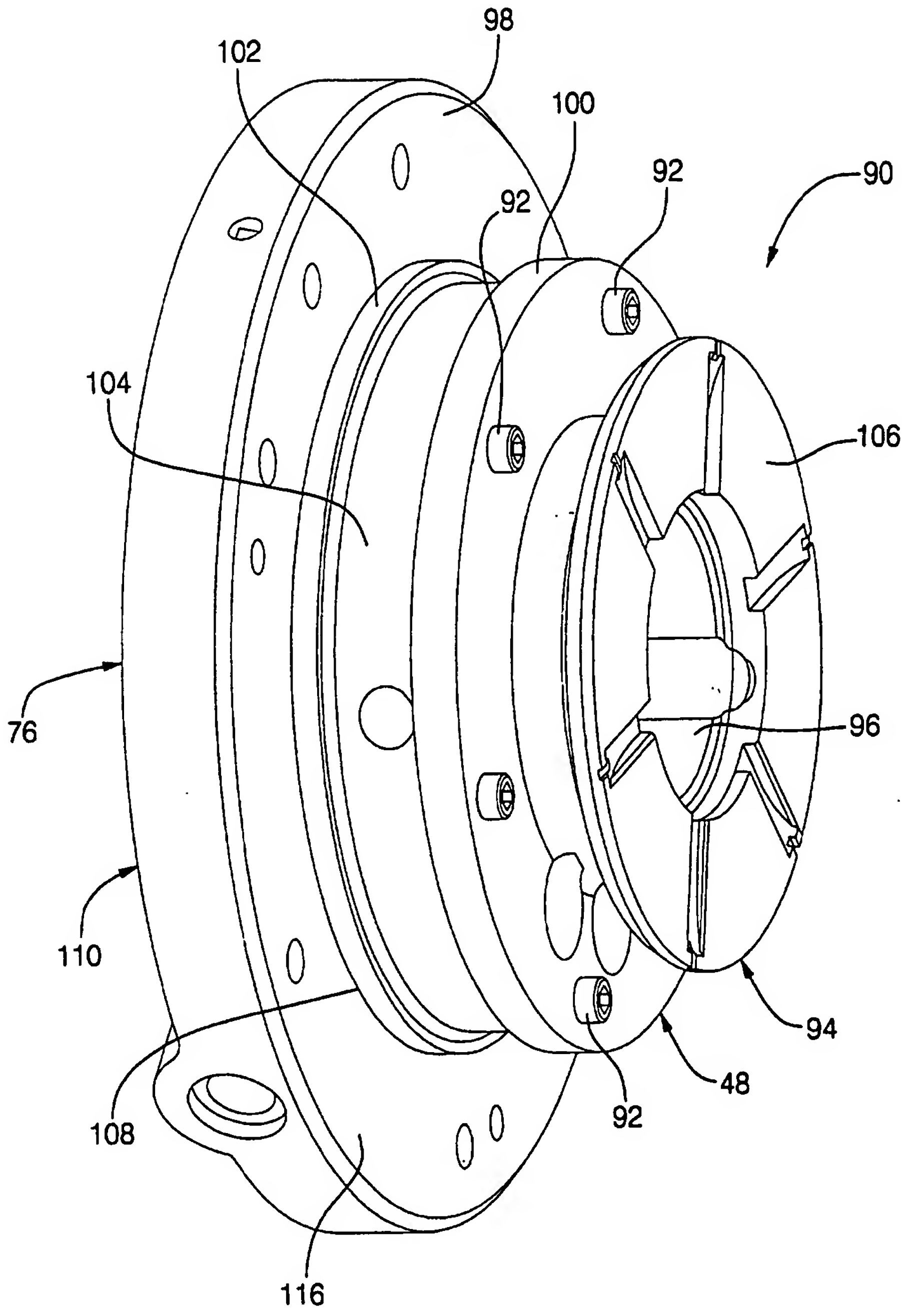

FIG 4

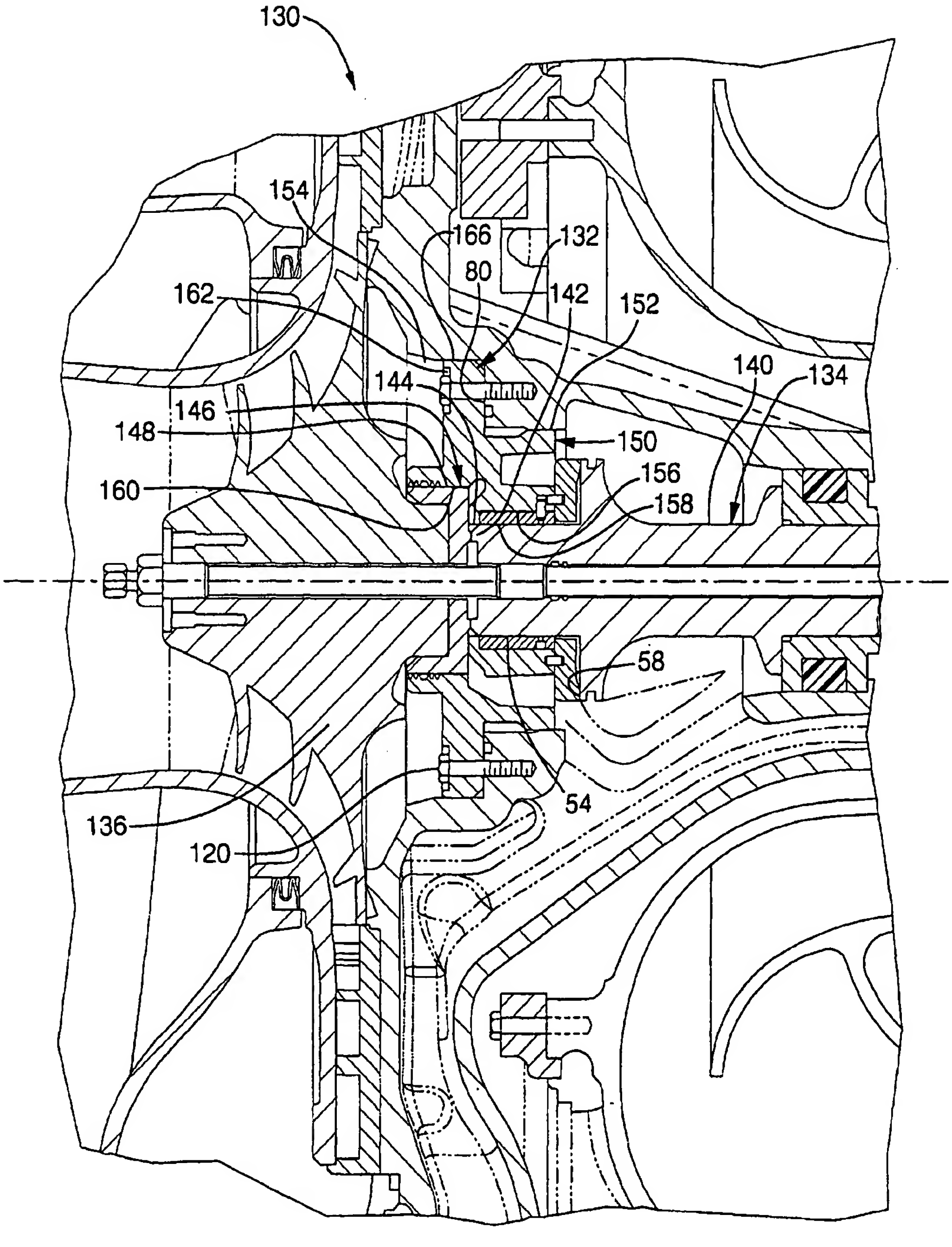
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FIG. 5

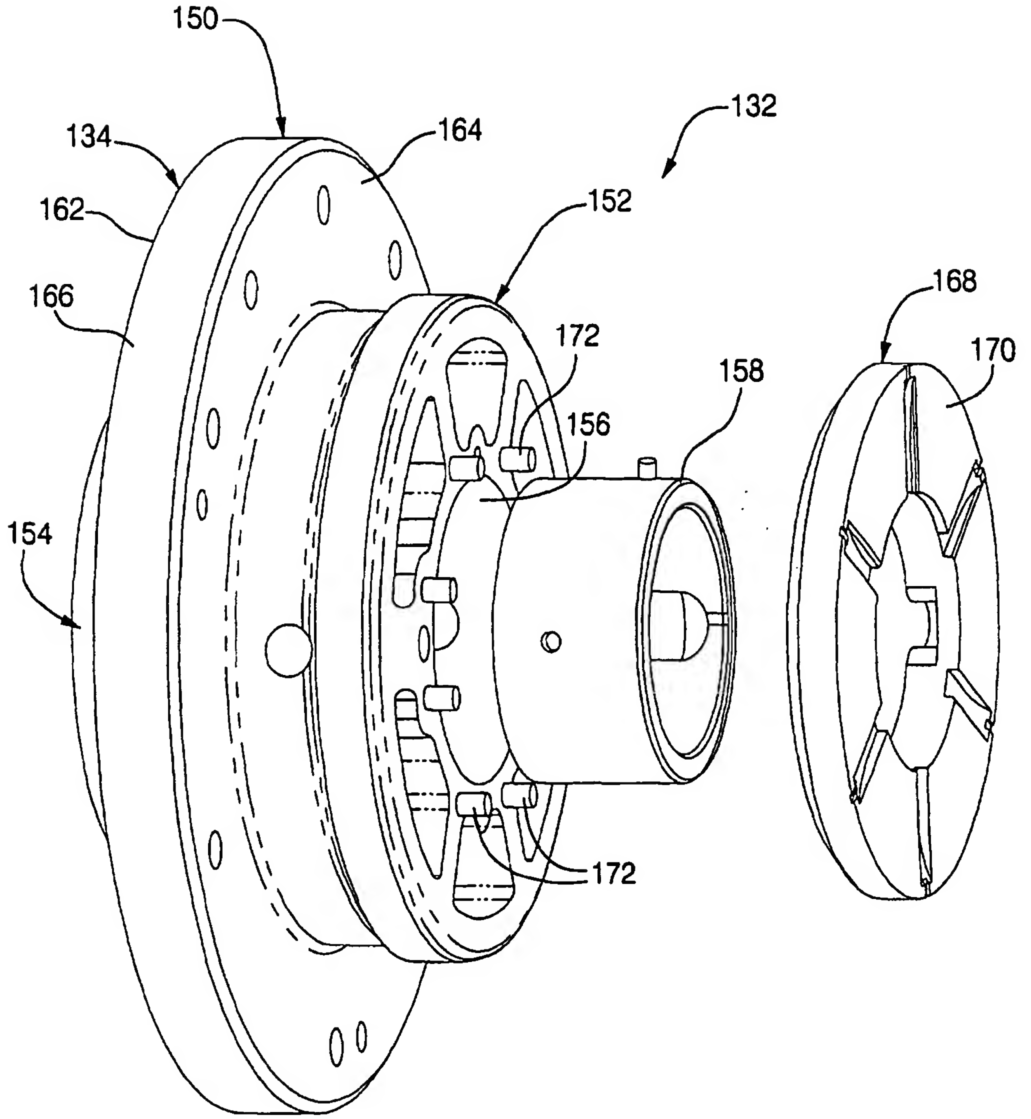

FIG. 6

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(54) Shaft seal arrangement of turbocharger.

(57) A shaft seal arrangement of a turbocharger (1) for an internal combustion engine having a throttle valve disposed upstream of a compressor of the turbocharger. First and second seal rings (12,13) are fitted on a cylindrical thrust spacer (11) fixed on a rotatable shaft (2) on which turbine and compressor wheels (4,3) are mounted. The seal rings (12,13) are spaced from each other and positioned between the compressor and bearings (6) for the shaft (2). An annular air chamber 14 between the seal rings (12,13) is supplied with atmospheric air through an air supply passage (15). The cross-sectional area of the air supply passage (15) is such that a pressure difference between the air chamber (14) and the bearing side is prevented from exceeding an experimental pressure difference value at which oil leak from the bearing side to the compressor side begins, thereby effectively avoiding oil leakage even in an engine operating condition in which the throttle valve is fully closed.

EP 0 357 246 A2

SHAFT SEAL ARRANGEMENT OF TURBOCHARGER

This invention relates to improvements in a shaft seal arrangement for a turbocharger, and more particularly to such a shaft seal arrangement for preventing oil leak from a bearing side to a compressor side of the turbocharger.

A turbocharger installed to an internal combustion engine is usually provided with a shaft seal arrangement for preventing lubricating oil supplied to bearings from leaking out to the side of a compressor of the turbocharger. The bearings support a rotatable shaft on which turbine and compressor wheels are fixedly mounted. Such a shaft seal arrangement is disclosed, for example, in Japanese Utility Model Provisional Publication Nos. 60-116035 and 61-166134, in which the rotatable shaft of the turbocharger is provided with two seal rings for preventing oil leak. An air supply passage is formed to supply air into between the two seal rings. In this case, oil leak from the bearings is suppressed by cancelling a pressure difference between the seal rings and the bearing housing.

However, in case that a throttle valve of an engine is disposed upstream of the compressor of the turbocharger, a greater vacuum is developed in the compressor in an engine operating condition in which the throttle valve is closed. As a result, the amount of air sucked out from between the seal rings to the side of the compressor wheel increases, so that air pressure between the seal rings lowers due to a pressure drop of air flowing through the air supply passage. This causes lubricating oil to leak from the bearing housing to the compressor wheel side of the turbocharger.

It is an object of the present invention to provide an improved shaft seal arrangement of a turbocharger, by which lubricating oil is prevented from leaking out from the side of a bearing for a rotatable shaft of the turbocharger thereby to reduce lubricating oil consumption.

A shaft seal arrangement of the present invention is of a turbocharger for an internal combustion engine having a throttle valve disposed upstream of a compressor of the turbocharger. First and second seal rings are disposed around and securely connected to a rotatable shaft of the compressor and located between the compressor and a bearing for the rotatable shaft. The first and second seal rings are located at the side of the bearing and the side of the compressor, respectively, and adapted to prevent oil leak from the side of the bearing to the side of the compressor. An air supply passage is such provided that a location between the first and second seal rings is supplied with atmospheric air through the air supply passage. The cross-sectional area A of the air supply passage is set to meet a relationship represented by the following equation under a condition of a temperature of 15°C and of standard atmospheric pressure:

$$A \geqq S \times \sqrt{\frac{K \times \gamma \times R \times \left(\frac{2}{K+1}\right)^{(K+1)/(K-1)}}{2 \times \Delta Pmin}}$$

where $\Delta Pmin$ is the minimum seal performance of the first seal ring; S is the maximum clearance of the seal ring; K is ratio of specific heats; R is the gas constant of air; and γ is the specific gravity of air.

With the above arrangement, a sufficient amount of air can be supplied to the location between the first and second seal rings even under an operating condition in which air in the above-mentioned location is sucked out to the compressor side through the clearance of the seal ring. This avoids that the pressure difference between the above-mentioned location and the bearing side (at atmospheric pressure) increases over the minimum seal performance $\Delta Pmin$ (or an experimental pressure difference value at which oil leak from the bearing side to the compressor side begins), thereby preventing oil leak from the bearing side to the compressor side. This largely suppresses oil consumption while preventing generation of white smoke in exhaust gas.

In the drawings, like reference numerals designate corresponding elements and parts throughout all figures, in which:

Fig. 1 is a longitudinal sectional view of a turbocharger including a first embodiment of a shaft seal arrangement according to the present invention;

Fig. 2 is an enlarged fragmentary sectional view of an essential part of the shaft seal arrangement of Fig. 1;

Fig. 3 is a fragmentary sectional view of a turbocharger including a second embodiment of the shaft seal arrangement according to the present invention;

Fig. 4 is an enlarged sectional view of the shaft seal arrangement of Fig. 3;

Fig. 5 is an enlarged sectional view similar to Fig. 4 but showing a modified example of the shaft seal arrangement of Figs. 3 and 4; and

Fig. 6 is a fragmentary sectional view similar to Fig. 3 but showing another modified example of the shaft seal arrangement of Figs. 3 and 4.

Referring now to Figs. 1 and 2, there is shown a first embodiment of a shaft seal arrangement of a turbocharger 1, in accordance with the present invention. The turbocharger 1 of this instance is of an automotive vehicle and includes a rotatable shaft 2 on which a compressor wheel 3 and a turbine wheel 4 are coaxially fixedly mounted spaced from each other. The compressor wheel 3 forms part of a compressor (no numeral) for intake air. The turbine wheel 4 forms part of a turbine for taking out a rotational force from exhaust gas. The rotatable shaft 2 is journalled through a floating metal 6 on a bearing housing 5. The floating metal 6 is maintained in a floating state relative to the inner supporting surface of the bearing housing 5 under the influence of engine lubricating oil supplied through an oil passage 7 leading from an engine oil lubricating system (not shown). A compressor housing 8 surrounding the compressor wheel 3 is connected through a back plate 9 to the bearing housing 5. A cylindrical thrust spacer 11 is mounted or fitted on the rotatable shaft 2 and rotatably fitted in a bore (no numeral) of the back plate 9. Two seal rings 12, 13 are generally coaxially interposed between the thrust spacer 11 and the back plate 9 thereby to accomplish fluid-tight seal between the bearing housing 5 and the compressor housing 8. More specifically, each seal ring 12, 13 is disposed in an annular groove (21 in Fig. 2) formed coaxially on the peripheral surface of the thrust spacer 11 as shown in Fig. 2, in which the outer periphery of each seal ring 12, 13 is contactable with the inner peripheral surface (defining the bore) of the back plate 9.

An annular air chamber 14 is formed to surround the peripheral surface of a longitudinally intermediate part of the thrust spacer 11. The air chamber 14 is communicated through an air supply passage 15 with an air intake passageway 17 upstream of a throttle valve 16. The air intake passageway 17 is fluidly connected through the compressor wheel 3 to engine cylinders (not shown) of the engine. The throttle valve 16 is rotatably disposed in the intake air passageway 17 upstream of the compressor wheel 3 to control air amount to be supplied to the engine cylinders.

In the above arrangement, the volume flow rate Q of air sucked from the air chamber 14 to the side of the compressor wheel 3 is smaller than a so-called choke flow rate or a flow rate of air flowing when the throttle valve 16 is fully closed. Therefore, the volume flow rate Q for a maximum clearance area S for the seal ring 13 under a standard condition of atmospheric pressure (P_0) and a temperature of 15°C is given by the following equation:

$$Q = S \times \sqrt{K \times g \times R \times \left(\frac{2}{K+1}\right)^{(K+1)/(K-1)}} \quad .. \quad (1)$$

where K is ratio of specific heats; g is gravitational acceleration; and R is the gas constant of air. The maximum clearance area S is determined by an equation of $S = \min(x_1, x_2) \times 2\pi r$ where x_1 and x_2 are respectively the opposite side clearances of the seal ring 13 relative to the wall surfaces of the groove 21; and r is the inner diameter of the seal ring 13 as shown in Fig. 2.

A pressure difference ΔP between the bearing housing 5 and the air chamber 14 must be smaller than a minimum seal performance $\Delta P\text{min}$. Here, the flow velocity v of air flowing through the air supply passage 15 in the standard condition is given by the following equation:

$$v = \sqrt{\frac{2g \times \Delta P}{\gamma}} \leqq \frac{2g \times \Delta P\text{min}}{\gamma} \quad .... \quad (2)$$

where γ is the specific gravity of air.

In the air supply passage 15, air must flow at this flow velocity and in the above-mentioned flow rate Q, and therefore the cross-sectional area A of a through-hole 18 forming part of the air flow passage 15 should be in a relationship as represented by the following equation derived from the above-equations (1) and (2):

$$A = \frac{Q}{V} \geq \frac{Q}{\sqrt{\frac{2g \times \Delta Pmin}{\gamma}}}$$

$$= S \times \sqrt{\frac{K \times \gamma \times R \times \left(\frac{2}{K+1}\right)^{(K+1)/(K-1)}}{2 \times \Delta Pmin}} \quad .. (3)$$

where K is ratio of specific heats; R is the gas constant of air; γ is the specific gravity of air. The minimum seal performance is determined by the following experiment: An experimental device is arranged in the same condition as in the turbocharger 1 shown in Fig. 1. The rotatable shaft (2) of the experimental device is rotated under a condition in which oil mist is filled on the side of the bearing housing (5), during which the pressure difference between the opposite positions of the ring seals is altered. A pressure value of the pressure difference at which oil mist begins to leak through the seal rings is determined as the minimum seal performance ΔPmin.

The operation of the thus arranged intake system will be discussed hereinafter.

The inside of the bearing housing 5 is maintained at about atmospheric pressure although there is a slight leak of exhaust gas from the side of the turbine wheel 4 into the bearing housing 5. The air chamber 14 is also maintained at about atmospheric pressure because of being supplied with atmospheric pressure prevailing in the air intake passageway 17 upstream of the throttle valve 16. In an engine operating condition in which the throttle valve 16 is closed, the back side of the compressor wheel 3 is under vacuum and therefore air is sucked out from the air chamber 14 to the side of the compressor wheel 3. The amount of the thus sucked out air is considerably small and therefore negligible. At this time, if the amount of air flowing through the air supply passage 15 becomes larger over a predetermined level, the air chamber 14 becomes under vacuum owing to a pressure drop in the air supply passage 15. This seems to cause oil leak through the seal ring 12 from the bearing housing 5.

However, according to this embodiment, the cross-sectional area of the air supply passage 15 is such determined that the pressure difference between the air chamber 14 and the bearing housing 5 does not lower below the minimum seal performance ΔPmin. Accordingly, the air chamber 14 is prevented from becoming under vacuum, thereby securely avoiding oil leak from the bearing housing 5.

Figs. 3 and 4 illustrate a second embodiment of the shaft seal arrangement in accordance with the present invention, which is similar to the first embodiment shaft seal arrangement. In this embodiment, the two seal rings 12, 13 are disposed between the cylindrical thrust spacer 11 and the back plate 9. More specifically, the seal rings 12, 13 are respectively fitted in the two annular grooves 21A, 21 formed coaxially on the outer peripheral surface of the thrust spacer 11, and contactable at its outer peripheral surface with the surface 38 of Jo the bore of the back plate 9. The annular air chamber 14 is defined between the seal rings 12, 13 and between the thrust spacer 11 and the back plate 9. The air chamber 13 is opened to atmospheric air through the through-hole 18 and the air supply passage 15.

The rotatable shaft 2 is journalled in the bearing housing 5 through a bearing metal 29 which is supplied with lubricating oil through a lubricating oil passage 27. The lubricating oil circulates through a bearing chamber B. A thrust collar 28 is disposed between the bearing metal 29 and the thrust spacer 11.

In this embodiment, the surface 38 of the bore of the back plate 9 is formed right cylindrical so that the seal rings 12, 13 are axially slidably supported to the bore surface 38. The back plate 9 is integrally formed with an annular stop section 39 which extends radially inwardly from the bore surface 38. The annular stop section 39 is formed on the side of the compressor wheel 3. The seal ring 13 at the compressor wheel side is contactable at its side surface with the annular stop section 39, thereby restricting the displacement of the seal ring 13 toward the compressor wheel 3.

Each seal ring 12, 13 is fitted in the annular groove 21A, 21 with a predetermined side clearance (a total clearance between the side surface of the seal ring and the side surface of the annular groove). The side clearance for the annular groove 21A and seal ring 12 at the bearing metal side is set at about 50 µm taking only machining error into consideration. The side clearance for the annular groove 13 and for the seal ring 21 on the compressor wheel side is set at a range of from 90 to 150 µm which is not less than 1.8 times of the above side clearance of about 50 µm, taking account of machining error and the axial displacement of the rotatable shaft 2.

In operation, the seal ring 32 seems to displace toward the side of the compressor wheel 3 under the action of a pressure difference (about 600 mmHg in maximum) between the back side A of the compressor wheel 3 and the bearing chamber B. However, the seal ring 13 is brought into contact with the annular stop

39 and therefore is prevented from displacement toward the compressor wheel 3. As a result, the seal ring 13 can be prevented from being pressed against the side surface of the goove 21. This prevents wear of seal ring 13 while suppressing the friction loss of the rotatable shaft 12 to a lower level, thus maintaining a high acceleration response of the turbocharger.

The annular air chamber 13 is supplied with atmosphric air and therefore always maintained at about atmospheric pressure regardless of engine operating conditions. Consequently, a pressure difference is hardly developed between the annular chamber 13 and the bearing chamber B so that no pressure difference is applied to the seal ring 12 on the bearing metal side. As a result, a force for pressing the seal ring 12 against the side wall of the groove 21A is very small, and accordingly wear of the seal ring 12 can be prevented though there is no annular stop section for the seal ring 12. This suppresses the friction loss of the rotatable shaft 2 at a lower value, thereby maintaining a high acceleration response of the turbocharger.

Since no step section is formed at the bore surface 38 of the back plate 9, the seal ring 12 is axially slidably supported and therefore the side clearance for the seal ring 12 and the annular groove 21A is not required to be set larger taking account of the axial displacement amount (about 40 to 100 µm) of the rotatable shaft 2. Consequently, the side clearance for the seal ring 12 is set, for example, at 50 µm taking account of only machining accuracy of annular groove 21A. As a result, the amount of lubricating oil leaking through the clearance between the seal ring 12 and the side surfaces of the annular groove 21A is minimized thereby providing a sufficient sealing ability.

The back plate 9 is formed with only one annular stop section (a step section) which requires a relatively high machining accuracy, thereby lowering production cost of the turbocharger. Besides, since only an annular tapered section 40 at an end section of the back plate bore surface 38 is required, the axial length of the rotatable shaft required for shaft seal is shortened and therefore the flexural rigidity of the rotatable shaft 2 is increased, thereby improving the vibrational characteristics and durability of the rotatable shaft 2.

Fig. 5 shows a modified example of the second example shaft seal arrangement. In this example, a further annular groove 46 is formed at the outer peripheral surface of the thrust spacer 43 and located between the annular groove 21A, 21. The annular groove 46 forms part of the annular air chamber 13, thereby increasing the volume of the air chamber 13 formed between the thrust spacer 11 and the back plate 9. By virtue of this air chamber 13 having an increased volume, a pressure variation within the annular air chamber 13 is suppressed thereby providing a stable sealing ability. Besides, the annular groove 46 decreases a rotating mass of the turbocharger and does not obstruct the installation operation of the seal ring 12 as compared with a case in which the corresponding annular groove (46) is formed in the back plate (37).

Fig. 6 shows another example of the second embodiment shaft seal arrangement in accordance with the present invention. In this example, an annular stopper plate 52 is installed to the back plate 9 at the end face 57 opposite to the compressor wheel 3 by means of small screws 53. The outer peripheral section of the annular stopper plate 52 extends radially inwardly from the bore surface 38 of the back plate 9 and therefore serves as the annular stop section (39) for resticting the movement of the seal ring 13. In this example, the back plate 9 is formed at its end surface opposite to the compressor wheel 3 with an annular groove 55 in which an annular projection 56 of the stopper plate 52 is fitted, so that an installation accuracy of the stopper plate 52 is improved. It will be understood that installation of the annular projection 56 of the stopper plate 52 may be made by means of either one of the press-fitting and the small screws 53. With this arrangement, locationing accuracy of the ring seal 13 is controlled by the side end face 57 of the back plate 9 thereby to lower a production cost of the turbocharger.

Claims

1. A shaft seal arrangement of a turbocharger (1) for an internal combustion engine having a throttle valve disposed upstream of a compressor of the turbocharger, the shaft seal arrangement comprising:
first and second seal rings (12,13) disposed around and securely connected to a rotatable shaft (2) of the compressor and located between the compressor and a bearing (6) for the rotatable shaft (2), the first and second seal rings (12,13) being located at the bearing side and the compressor side, respectively, and adapted to prevent oil leakage from the bearing side to the compressor side; and
an air supply passage (15) through which a location between the seal rings (12,13) is supplied with substantially atmospheric pressure;
the cross-sectional area A of the air supply passage (15) being set to meet a relationship represented by

the following equation under the condition of a temperature of 15°C and of standard atmospheric pressure:

$$A \geqq S \times \sqrt{\frac{K \times \gamma \times R \times \left(\frac{2}{K+1}\right)^{(K+1)/(K-1)}}{2 \times \Delta Pmin}}$$

where Pmin is the minimum seal performance of the first seal ring (12), S is the maximum clearance of the second seal ring (13), K is ratio of specific heats; R is the gas constant of air, and γ is the specific gravity of air.

2. A shaft seal arrangement as claimed in claim 1, including a cylindrical thrust spacer (11) fitted on the rotatable shaft (2) and rotatably supported by a fixed support member 9, the thrust spacer (11) being formed with first and second coaxial annular grooves in which the first and second seal rings (12,13) are disposed respectively, the said maximum clearance S being defined between the second seal ring (13) and a wall surface of the thrust spacer defining the first annular groove, the seal rings (12,13) being slidingly contactable with the fixed support member (11).

3. A shaft seal arrangement as claimed in claim 1 or 2, including means for maintaining the bearing (6) in a floating state under influence of oil, and means for supplying the said oil around the bearing.

4. A shaft seal arrangement as claimed in any of claims 1 to 3, including means defining a generally right cylindrical supporting surface on which the first and second seal rings (12,13) are slidably supported at their outer periphery, the said cylindrical supporting surface being formed around the rotatable shaft (2), and a stop section in fixed relation to the said cylindrical supporting surface defining means and located at the compressor side relative to the second seal ring (13), the stop section extending radially inwardly relative to the cylindrical supporting surface, said second seal ring being contactable with the stop section, and means defining first and second annular grooves provided on the side of the rotatable shaft (2), the first and second seal rings (12,13) being fitted in the respective first and second annular grooves with predetermined side clearances.

5. A shaft seal arrangement of a turbocharger (1) for an internal combustion engine having a throttle valve disposed upstream of a compressor of the turbocharger, the shaft seal arrangement comprising:
first and second seal rings (12,13) disposed around and securely connected to a rotatable shaft (2) of the compressor and located between the compressor and a bearing (6) for the rotatable shaft (2), the first and second seal rings (12,13) being located at the bearing side and the compressor side, respectively, and adapted to prevent oil leakage from the bearing side to the compressor side;
an air supply passage (15) through which a location between the seal rings (12,13) is supplied with substantially atmospheric pressure;
means defining a generally cylindrical supporting surface on which the seal rings (12,13) are slidably supported at their outer periphery, the cylindrical supporting surface being formed around the rotatable shaft (2);
a stop section in fixed relation to the cylindrical supporting surface defining means and located at the compressor side relative to the second seal ring (13), the stop section extending radially inwardly relative to the cylindrical supporting surface, the second seal ring (13) being contactable with the stop section; and
means defining first and second annular grooves provided at the side of the rotatable shaft (2), the first and second seal rings (12,13) being fitted in the respective first and second annular grooves with predetermined side clearances.

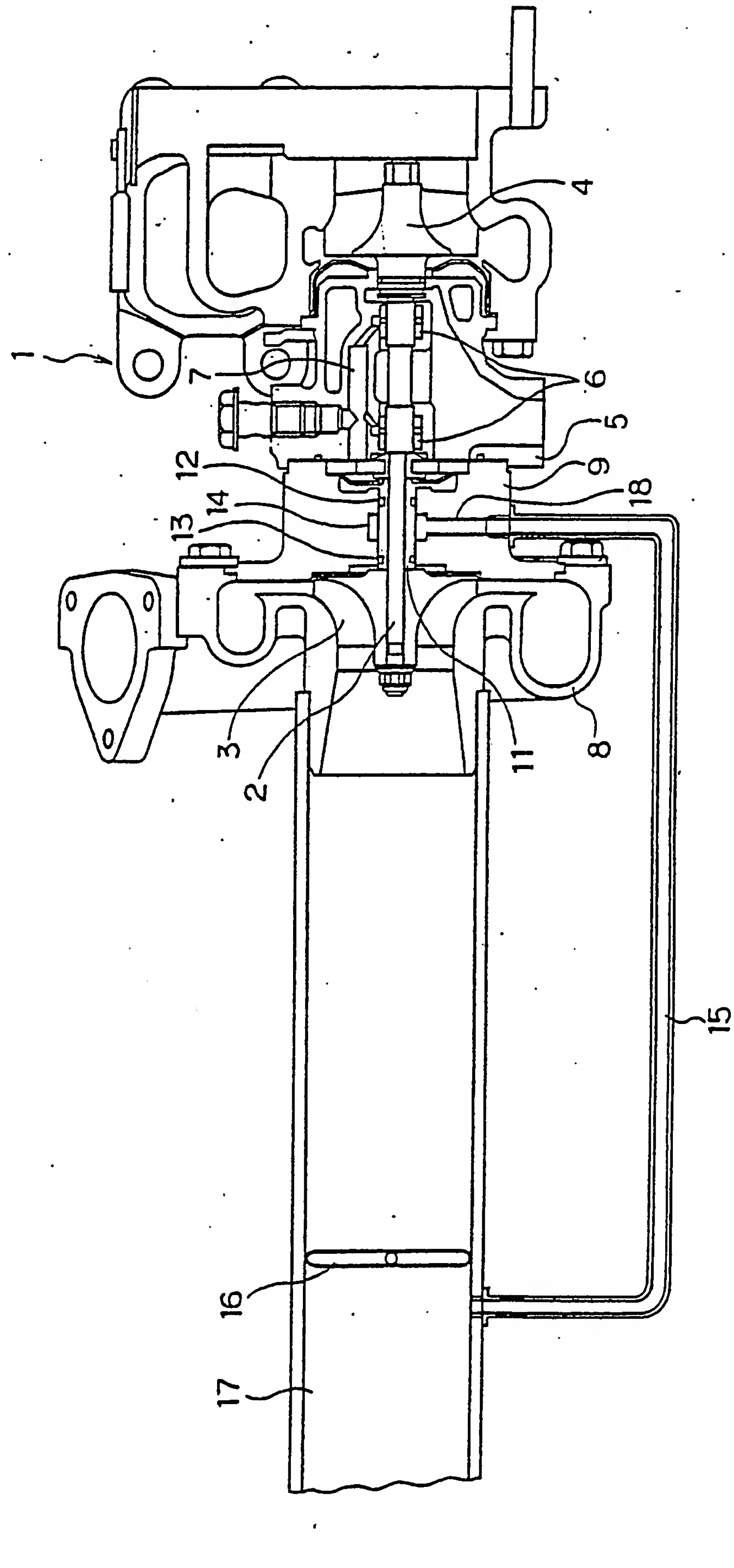


FIG. 1

FIG. 2

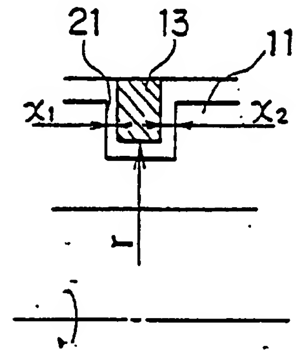


FIG. 3

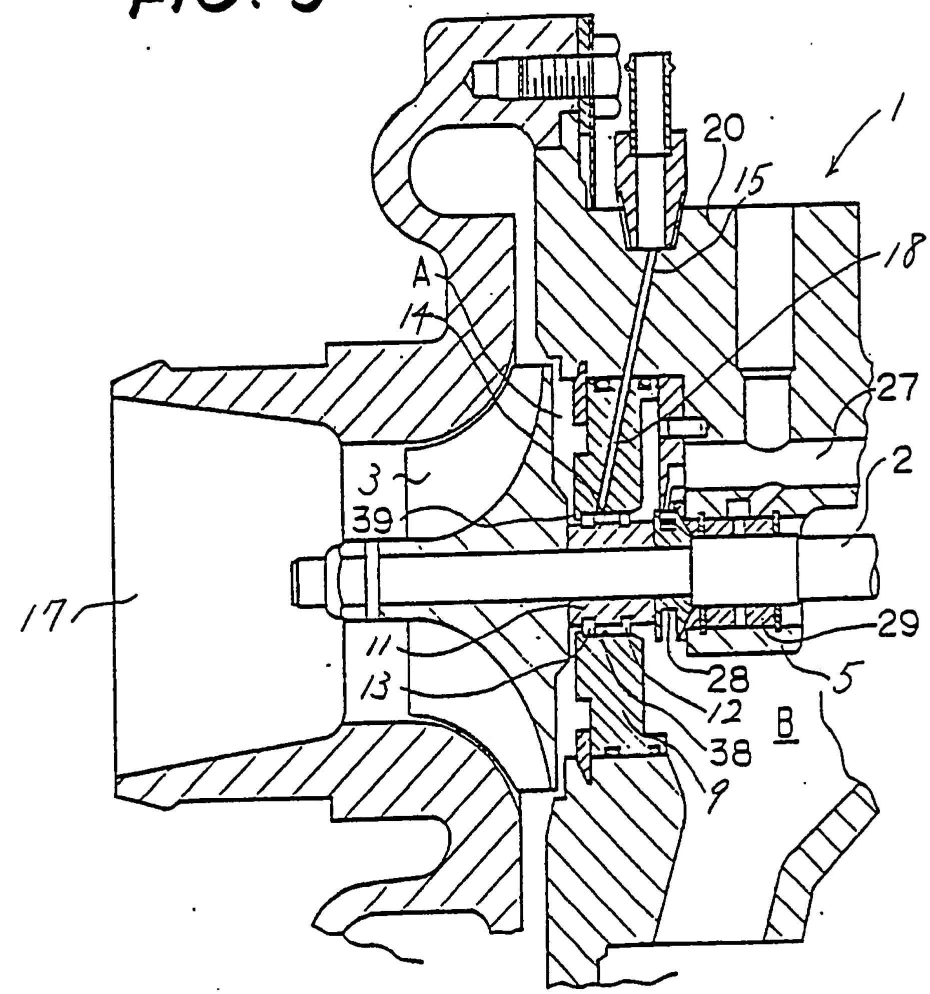


FIG. 4

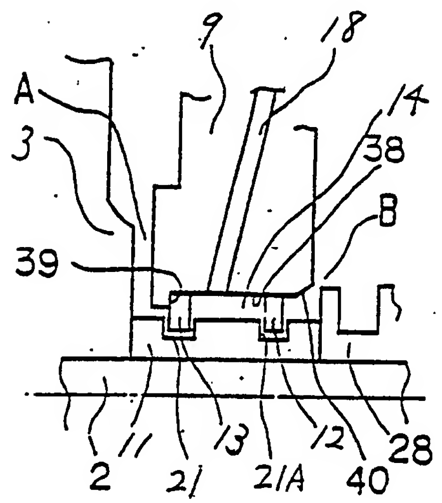


FIG. 5

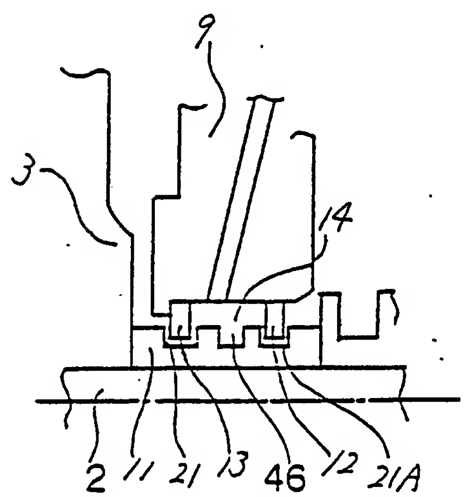
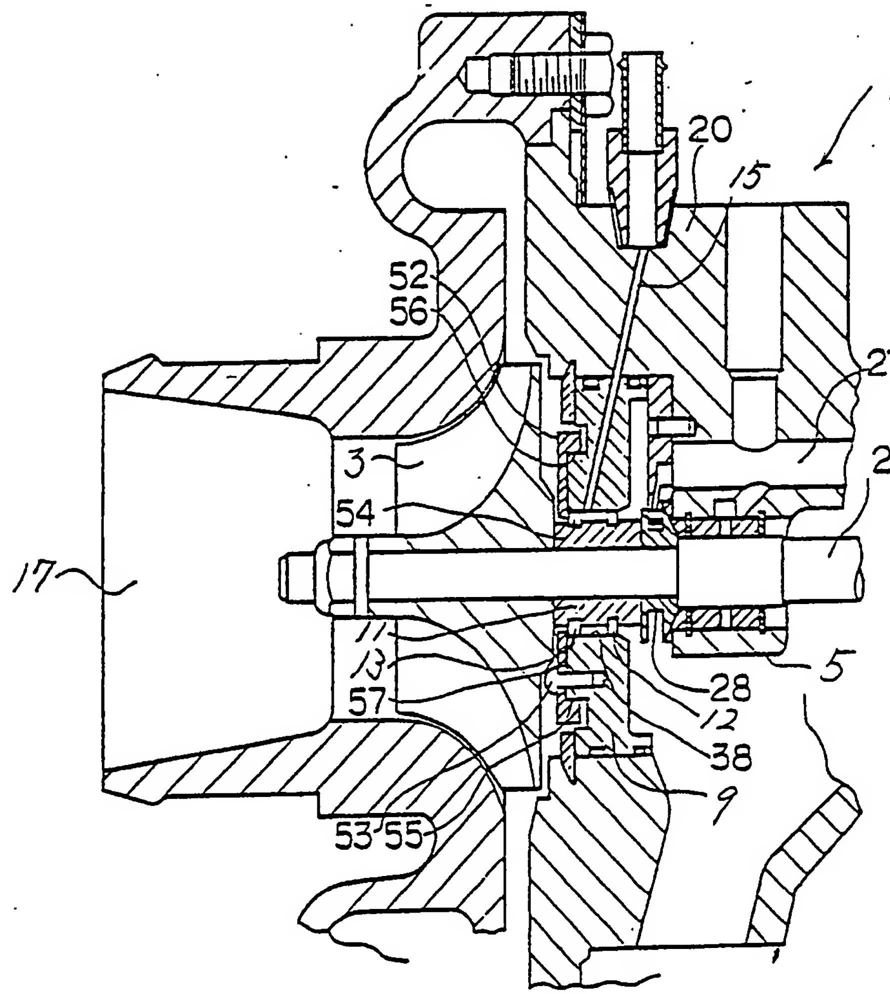


FIG. 6



PATENT COOPERATION TREATY

PCT

INTERNATIONAL SEARCH REPORT

(PCT Article 18 and Rules 43 and 44)

Applicant's or agent's file reference DKT 05106A (BWI-0015	**FOR FURTHER ACTION**	see Form PCT/ISA/220 as well as, where applicable, item 5 below.
International application No. PCT/US2006/033783	International filing date *(day/month/year)* 29/08/2006	(Earliest) Priority Date *(day/month/year)* 09/09/2005
Applicant BORGWARNER INC.		

This international search report has been prepared by this International Searching Authority and is transmitted to the applicant according to Article 18. A copy is being transmitted to the International Bureau.

This international search report consists of a total of 4 sheets:

[X] It is also accompanied by a copy of each prior art document cited in this report.

1. **Basis of the report**
   a. With regard to the **language**, the international search was carried out on the basis of:
      [X] the international application in the language in which it was filed
      [ ] a translation of the international application into ____________, which is the language of a translation furnished for the purposes of international search (Rules 12.3(a) and 23.1(b))
   b. [ ] With regard to any **nucleotide and/or amino acid sequence** disclosed in the international application, see Box No. I.

2. [ ] **Certain claims were found unsearchable** (See Box No. II)

3. [ ] **Unity of invention is lacking** (see Box No III)

4. With regard to the **title**,
   [ ] the text is approved as submitted by the applicant
   [X] the text has been established by this Authority to read as follows:

TURBOCHARGER WITH BEARING HOUSING HAVING AN AERODYNAMICALLY ENHANCED COMPRESSOR WHEEL POCKET GEOMETRY

5. With regard to the **abstract**,
   [X] the text is approved as submitted by the applicant
   [ ] the text has been established, according to Rule 38.2(b), by this Authority as it appears in Box No. IV. The applicant may, within one month from the date of mailing of this international search report, submit comments to this Authority

6. With regard to the **drawings**,
   a. the figure of the **drawings** to be published with the abstract is Figure No. 1, 2
      [ ] as suggested by the applicant
      [X] as selected by this Authority, because the applicant failed to suggest a figure
      [ ] as selected by this Authority, because this figure better characterizes the invention
   b. [ ] none of the figures is to be published with the abstract

INTERNATIONAL SEARCH REPORT

International application No
PCT/US2006/033783

A. CLASSIFICATION OF SUBJECT MATTER
INV. F04D29/08 F01D25/18

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F02C F01D F04D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	EP 1 193 372 A (GEN MOTORS CORP [US] ELECTRO MOTIVE DIESEL INC [US]) 3 April 2002 (2002-04-03) column 3, paragraph 9 column 4, lines 35-40 column 4, paragraphs 15,16 figures 1,2 -----	1,4-11, 13-19, 22,23
X	US 6 418 722 B1 (ARNOLD STEVEN DON [US]) 16 July 2002 (2002-07-16) column 2, lines 23-47 figure 1 -----	1-3, 7-10,12, 14-21
	-/--	

[X] Further documents are listed in the continuation of Box C.

[X] See patent family annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
23 January 2007	01/02/2007

Name and mailing address of the ISA/	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31–70) 340–2040, Tx. 31 651 epo nl, Fax: (+31–70) 340–3016	Souris, Christophe

3

C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	EP 0 357 246 A (NISSAN MOTOR [JP]) 7 March 1990 (1990-03-07) abstract page 2, lines 4-46 page 3, lines 42-44 page 4, lines 17-31 figure 3	1,7,14
A	US 2005/058533 A1 (BELOKON ALEXANDER [RU] ET AL) 17 March 2005 (2005-03-17) page 1, paragraph 3 page 1, paragraphs 6,7 page 3, paragraph 26 figure 1	1,7,14
A	US 1 488 931 A (CHARLES MARECHAL PAUL JOSEPH) 1 April 1924 (1924-04-01) page 1, lines 30-38,61-107 page 2, lines 5-40 figure 1	1,7,14

3

INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No
PCT/US2006/033783

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